15\PRTS

10/516883
DT05 Rec'd PCT/PTO 03 DEC 2004

DESCRIPTION

EXPANDER

FIELD OF THE INVENTION

The present invention relates to an expander in which an axial piston cylinder group is arranged annularly in a rotor so as to surround an axis of the rotor, the rotor being rotatably supported in a casing, and spherical projections formed at the forward end of pistons of the axial piston cylinder group abut against spherical depressions formed in a swash plate.

BACKGROUND ART

A hydraulic device in which a spherical projection formed at the forward end of a piston of an axial piston cylinder group abuts against a spherical depression formed in a swash plate is known from Japanese Patent Application Laid-open No. 61-274166. By making the spherical projection abut against the spherical depression it is possible to reduce the surface pressure in the area where the two abut against each other and prevent relative rotation between the swash plate and the piston, thereby contributing to a reduction in wear of the swash plate and the piston. Moreover, since the piston exerts an aligning action on the swash plate, it is possible to lighten the load imposed on a swash plate holder supporting the swash plate, thereby improving the durability.

However, since the axis of the swash plate is inclined relative to the axis of a rotor supporting the axial piston cylinder group, even if a plurality of spherical depressions are arranged in the peripheral direction around the axis of the swash plate, the locus of the contact points between the plurality of piston spherical projections and the spherical depressions is elliptical. That is, the contact points between the spherical projections of the pistons and the spherical depressions of the swash plate are eccentric with respect to the axis

of the piston and the axis of the spherical depression, and the direction of the load imposed on the spherical projection of the piston by the spherical depression of the swash plate deviates from the direction of the piston axis. As a result, the piston receives a bending moment or a radially biased load, and the bending moment or radially biased load causes galling on sliding faces of the piston and the cylinder, thus giving rise to the problems of increased sliding resistance and the occurrence of abnormal wear.

DISCLOSURE OF INVENTION

The present invention has been achieved under the above-mentioned circumstances, and it is an object thereof to reduce the bending moment or the radially biased load exerted on the piston of the expander by the swash plate, thereby minimizing any increase in sliding resistance and any occurrence of abnormal wear.

In order to attain this object, in accordance with a first aspect of the present invention, there is proposed an expander that includes a casing, a rotor rotatably supported in the casing, an axial piston cylinder group arranged annularly in the rotor so as to surround an axis of the rotor, and a swash plate rotatably supported in the casing so that an axis of the swash plate is inclined at a predetermined angle relative to the axis of the rotor, spherical projections formed at the forward end of pistons of the axial piston cylinder group abutting against spherical depressions formed in the swash plate so as to coaxially surround the axis of rotation of the swash plate, and the rotor being rotated by supplying via a rotary valve high-temperature, high-pressure steam to expansion chambers defined between the pistons of the axial piston cylinder group and cylinder sleeves, characterized in that a locus of contact points between the spherical depressions of the swash plate and the spherical

projections of the piston is offset toward the expansion stroke side of the axial piston cylinder group.

In accordance with this arrangement, with regard to the expander that includes the axial piston cylinder group, since the locus of the contact points between the spherical depressions of the swash plate and the spherical projections of the pistons is offset toward the expansion stroke side of the axial piston cylinder group, in a middle region of the expansion stroke where the speed of the piston is high and the surface pressure at the contact point between the piston and the swash plate is large, the position of the contact point between the spherical depression of the swash plate and the spherical projection of the piston is made to be as close as possible to an axis of the spherical depression and an axis of the piston, thus reducing the bending moment and the radially biased load acting on the piston and thereby minimizing any increase in sliding resistance and any occurrence of abnormal wear.

Furthermore, in accordance with a second aspect of the present invention, in addition to the first aspect, there is proposed the expander wherein the axis of the swash plate is offset toward the exhaust stroke side of the axial piston cylinder group relative to the axis of the rotor.

In accordance with this arrangement, by employing a simple arrangement in which the axis of the swash plate is offset toward the exhaust stroke side of the axial piston cylinder group relative to the axis of the rotor, it is possible to offset the locus of the contact points between the spherical depressions of the swash plate and the spherical projections of the pistons toward the expansion stroke side of the axial piston cylinder group.

BRIEF DESCRIPTION OF DRAWINGS

FIG. 1 to FIG. 17 show one embodiment of the present invention; FIG. 1 is a vertical sectional view of an expander, FIG. 2 is a sectional view along line 2-2 in FIG. 1, FIG. 3 is a view from arrowed line 3-3 in FIG. 1, FIG. 4 is an enlarged view of part 4 in FIG. 1, FIG. 5 is an enlarged view of part 5 in FIG. 1, FIG. 6 is an exploded perspective view of a rotor, FIG. 7 is a sectional view along line 7-7 in FIG. 4, FIG. 8 is a sectional view along 8-8 in FIG. 4, FIG. 9 is an enlarged view of part 9 in FIG. 4, FIG. 10 is a sectional view along line 10-10 in FIG. 5, FIG. 11 is a sectional view along line 11-11 in FIG. 5, FIG. 12 is a sectional view along line 12-12 in FIG. 5, FIG. 13 is a sectional view along line 13-13 in FIG. 5, Fig. 14 is a view from arrowed line 14-14 in FIG. 4, Fig. 15 is a diagram for explaining the operation (in a case where there is offset), Fig. 16 is a diagram for explaining the operation (in a case where there is no offset), and Fig. 17 is a graph for explaining the effect of the offset.

BEST MODE FOR CARRYING OUT THE INVENTION

One embodiment of the present invention is explained below with reference to the attached drawings.

As shown in FIG. 1 to FIG. 9, an expander M of this embodiment is used in, for example, a Rankine cycle system, and converts the thermal energy and the pressure energy of high-temperature, high-pressure steam as a working medium into mechanical energy and outputs it. A casing 11 of the expander M is formed from a casing main body 12, a front cover 15 joined via a seal 13 to a front opening of the casing main body 12 by a plurality of bolts 14, a rear cover 18 joined via a seal 16 to a rear opening of the casing main body 12 by a plurality of bolts 17, and an oil pan 21 joined via a seal 19 to a lower opening of the casing main body 12 by a plurality of bolts 20.

A rotor 22 disposed rotatably around an axis L extending in the fore-and-aft direction in the center of the casing 11 has a front part thereof supported by a ball bearing 23 provided in the front cover 15 and a rear part thereof supported by a ball bearing 24 provided in the casing main body 12. A swash plate holder 28 is fitted in a rear face of the front cover 15 via two seals 25 and 26 and a knock pin 27, and fixed thereto via a plurality of bolts 29, and a swash plate 31 is rotatably supported by the swash plate holder 28 via an angular ball bearing 30. An axis L1 of the swash plate 31 is inclined relative to the axis L of the rotor 22, and the angle of inclination is fixed.

As is most clearly shown in Fig. 14, the axis L1 of the swash plate 31 is offset by a distance α toward the exhaust stroke side (left-hand side in the figure) of an axial piston cylinder group 56, which will be described later, relative to the axis L of the rotor 22.

The rotor 22 includes an output shaft 32 supported by the front cover 15 via the ball bearing 23, three sleeve support flanges 33, 34, and 35 formed integrally with a rear part of the output shaft 32 via cutouts 57 and 58 having predetermined widths (see FIG. 4 and FIG. 9), a rotor head 38 that is joined by a plurality of bolts 37 to the rear sleeve support flange 35 via a metal gasket 36 and supported by the casing main body 12 via the ball bearing 24, and a heat-insulating cover 40 that is fitted over the three sleeve support flanges 33, 34, and 35 from the front and joined to the front sleeve support flange 33 by a plurality of bolts 39.

Sets of five sleeve support holes 33a, 34a, and 35a are formed in the three sleeve support flanges 33, 34, and 35 respectively at intervals of 72° around the axis L, and five cylinder sleeves 41 are fitted into the sleeve support holes 33a, 34a, and 35a from the rear. A flange 41a is formed on the rear end of each of the cylinder sleeves 41, and axial positioning is carried out by

abutting this flange 41a against the metal gasket 36 while fitting the flange 41a into a step 35b formed in the sleeve support holes 35a of the rear sleeve support flange 35 (see FIG. 9). A piston 42 is slidably fitted within each of the cylinder sleeves 41, a spherical depression 61a at the front end of the piston 42 abutting against a spherical depression 31a, which is a dimple formed on the swash plate 31, and a steam expansion chamber 43 is defined between the rear end of the piston 42 and the rotor head 38.

An oil passage 32a is formed so as to extend on the axis L within the output shaft 32, which is integral with the rotor 22, and the front end of the oil passage 32a branches in a radial direction and communicates with an annular channel 32b on the outer periphery of the output shaft 32. An oil passage blocking member 45 is screwed into the inner periphery of the oil passage 32a via a seal 44 at a radially inner position of the middle sleeve support flange 34 of the rotor 22, and a plurality of oil holes 32c extending radially outward from the oil passage 32a in the vicinity of the oil passage blocking member 45 open on the outer periphery of the output shaft 32.

A trochoidal oil pump 49 is disposed between a recess 15a provided in a front face of the front cover 15 and a pump cover 48 fixed via a seal 46 to the front face of the front cover 15 by a plurality of bolts 47, and includes an outer rotor 50 that is rotatably fitted in the recess 15a, and an inner rotor 51 that is fixed to the outer periphery of the output shaft 32 and meshes with the outer rotor 50. An internal space of the oil pan 21 communicates with an intake port 53 of the oil pump 49 via an oil pipe 52 and an oil passage 15b of the front cover 15, and a discharge port 54 of the oil pump 49 communicates with the annular channel 32b of the output shaft 32 via an oil passage 15c of the front cover 15.

The piston 42, which is slidably fitted into the cylinder sleeve 41, is formed from an end part 61, a middle part 62, and a top part 63. The end part 61 is a member having the spherical projection 61a that abuts against the spherical depression 31a of the swash plate 31, and is joined by welding to the forward end of the middle part 62. The middle part 62 is a cylindrical member having a large volume hollow space 62a; an outer peripheral part of the middle part 62 close to the top part 63 has a small diameter part 62b whose diameter is slightly reduced, a plurality of oil holes 62c are formed so as to run radially through the small diameter part 62b, and a plurality of spiral oil channels 62d are formed in an outer peripheral part that is present forward of the small diameter part 62b. The top part 63 faces the expansion chamber 43 and is formed integrally with the middle part 62, and a heat-insulating space 65 is formed between a dividing wall 63a formed on an inner face of the top part 63 and a cover member 64 fitted into and welded to a rear end face of the top part 63 (see FIG. 9). Two compression rings 66 and one oil ring 67 are mounted on the outer periphery of the top part 63, and an oil ring channel 63b into which the oil ring 67 is fitted communicates with the hollow space 62a of the middle part 62 via a plurality of oil holes 63c.

The end part 61 and the middle part 62 of the piston are made of high-carbon steel, and the top part 63 is made of stainless steel; among these, the end part 61 is subjected to induction hardening, whereas the middle part 62 is subjected to hardening. As a result, high surface pressure resistance can be imparted to the end part 61, which abuts against the swash plate 31 at a high surface pressure, abrasion resistance can be imparted to the middle part 62, which is in sliding contact with the cylinder sleeve 41 under severe lubrication conditions, and heat resistance and corrosion resistance can be imparted to the

top part 63, which faces the expansion chamber 43 and is exposed to high temperature and high pressure.

An annular channel 41b is formed on the outer periphery of a middle part of the cylinder sleeve 41 (see FIG. 6 and FIG. 9), and a plurality of oil holes 41c are formed in the annular channel 41b. Regardless of where rotationally the cylinder sleeve 41 is mounted, the oil holes 32c formed in the output shaft 32 and oil holes 34b formed in the middle sleeve support flange 34 of the rotor 22 (see FIG. 4 and FIG. 6) communicate with the annular channel 41b. A space 68 formed between the heat-insulating cover 40 and the front and rear sleeve support flanges 33 and 35 of the rotor 22 communicates with the internal space of the casing 11 via oil holes 40a formed in the heat-insulating cover 40 (see FIG. 4 and FIG. 7).

An annular cover member 69 is welded to the front, or expansion chamber 43 side, of the rotor head 38, which is joined to the rear face of the front sleeve support flange 33 of the rotor 22 by the bolts 37, and an annular heat-insulating space 70 is defined at the back, or rear face of the cover member 69 (see FIG. 9). The rotor head 38 is positioned rotationally relative to the rear sleeve support flange 35 by a knock pin 55.

The five cylinder sleeves 41 and the five pistons 42 form an axial piston cylinder group 56 of the present invention.

The structure of a rotary valve 71 for the supply and discharge of steam to and from the five expansion chambers 43 of the rotor 22 is now explained with reference to FIG. 5, and FIG. 10 to FIG. 13.

As shown in FIG. 5, the rotary valve 71, which is disposed along the axis L of the rotor 22, includes a valve main body 72, a stationary valve plate 73, and a movable valve plate 74. The movable valve plate 74 is fixed to a rear face of the rotor 22 by a bolt 76 screwed into the oil passage blocking member

45 (see FIG. 4) while being positioned in the rotational direction by a knock pin 75. The bolt 76 also has the function of fixing the rotor head 38 to the output shaft 32.

As is clear from FIG. 5, the stationary valve plate 73, which abuts against the movable valve plate 74 via flat sliding surfaces 77, is fixed to the center of a front face of the valve main body 72 by one bolt 78, and also to an outer peripheral part of the valve main body 72 by an annular fixing ring 79 and a plurality of bolts 80. During this process, a step 79a formed on the inner periphery of the fixing ring 79 is press-fitted around the outer periphery of the stationary valve plate 73 so as to be fitted in a spigot joint manner, and a step 79b formed on the outer periphery of the fixing ring 79 is press-fitted in a spigot joint manner around the outer periphery of the valve main body 72, thereby ensuring that the stationary valve plate 73 is coaxial with the valve main body 72. A knock pin 81 is disposed between the valve main body 72 and the stationary valve plate 73, and determines the position of the stationary valve plate 73 in the rotational direction.

When the rotor 22 rotates, the movable valve plate 74 and the stationary valve plate 73 therefore rotate relative to each other on the sliding surfaces 77 in a state in which they are in intimate contact with each other. The stationary valve plate 73 and the movable valve plate 74 are made of a material having excellent durability, such as carbon or a ceramic, and the durability can be further improved by providing or coating the sliding surfaces 77 with a member having heat resistance, lubricating properties, corrosion resistance, and abrasion resistance.

The valve main body 72, which is made of stainless steel, is a stepped cylindrical member having a large diameter part 72a and a small diameter part 72b; outer peripheral faces of the large diameter part 72a and the small

diameter part 72b are slidably fitted in the axial L direction into circular cross-section support faces 18a and 18b of the rear cover 18 via seals 82 and 83 respectively, and positioned in the rotational direction by fitting a pin 84 implanted in an outer peripheral face of the valve main body 72 into a cutout 18c formed in the axial L direction in the rear cover 18. A plurality of preload springs 85 are supported in the rear cover 18 so as to surround the axis L, and the valve main body 72, which has a step 72c between the large diameter part 72a and the small diameter part 72b pushed by these preload springs 85, is biased forward so as to put the sliding surfaces 77 of the stationary valve plate 73 and the movable valve plate 74 in intimate contact.

A steam supply pipe 86 connected to a rear face of the valve main body 72 communicates with the sliding surfaces 77 via a first steam passage P1 formed in the interior of the valve main body 72 and a second steam passage P2 formed in the stationary valve plate 73. A steam discharge chamber 88 sealed by a seal 87 is formed between the casing main body 12, the rear cover 18, and the rotor 22, and this steam discharge chamber 88 communicates with the sliding surfaces 77 via sixth and seventh steam passages P6 and P7 formed in the interior of the valve main body 72 and a fifth steam passage P5 formed in the stationary valve plate 73. Provided on surfaces where the valve main body 72 and the stationary valve plate 73 are joined are a seal 89 surrounding a part where the first and second steam passages P1 and P2 are connected to each other and a seal 90 surrounding a part where the fifth and sixth steam passages P5 and P6 are connected to each other.

Five third steam passages P3 disposed at equal intervals so as to surround the axis L run through the movable valve plate 74, and opposite ends of five fourth steam passages P4 formed in the rotor 22 so as to surround the axis L communicate with the third steam passages P3 and the expansion

chambers 43. The part of the second steam passage P2 opening on the sliding surface 77 is circular, whereas the part of the fifth steam passage P5 opening on the sliding surface 77 has an arc shape with the axis L as its center.

The operation of the expander M of this embodiment having the above-mentioned arrangement is now explained.

High-temperature, high-pressure steam generated by heating water in an evaporator reaches the sliding surfaces 77 of the stationary valve plate 73 with the movable valve plate 74 from the steam supply pipe 86 via the first steam passage P1 formed in the valve main body 72 of the rotary valve 71 and the second steam passage P2 formed in the stationary valve plate 73, which is integral with the valve main body 72. The second steam passage P2 opening on the sliding surface 77 communicates momentarily during a predetermined intake period with the corresponding third steam passage P3 formed in the movable valve plate 74, which rotates integrally with the rotor 22, and the high-temperature, high-pressure steam is supplied, via the fourth steam passage P4 formed in the rotor 22, from the third steam passage P3 to the expansion chamber 43 within the cylinder sleeve 41.

Even after the communication between the second steam passage P2 and the third steam passage P3 has been blocked due to rotation of the rotor 22, the high-temperature, high-pressure steam expands within the expansion chamber 43 and causes the piston 42 fitted in the cylinder sleeve 41 to be pushed forward from top dead center toward bottom dead center, and the spherical projection 61a of the end part 61 at the front end of the piston 42 pushes against the spherical depression 31a of the swash plate 31. As a result, the reaction force that the piston 42 receives from the swash plate 31 gives a rotational torque to the rotor 22. For each one fifth of a revolution of the

rotor 22, the high-temperature, high-pressure steam is supplied into a fresh adjoining expansion chamber 43, thus continuously rotating the rotor 22.

While the piston 42, having reached bottom dead center accompanying rotation of the rotor 22, retreats toward top dead center by being pushed by the swash plate 31, the low-temperature, low-pressure steam pushed out of the expansion chamber 43 is discharged into the steam discharge chamber 88 via the fourth steam passage P4 of the rotor 22, the third steam passage P3 of the movable valve plate 74, the sliding surfaces 77, the arc-shaped fifth steam passage P5 of the stationary valve plate 73, and the sixth and seventh steam passages P6 and P7 of the valve main body 72, and is supplied therefrom into a condenser.

Since the axis L1 of the swash plate 31 is inclined relative to the axis L of the rotor 22, the locus (contact point locus T) of the contact points p between the spherical projections 61a of the pistons 42 and the spherical depressions 61a of the swash plate 31 is elliptical. Fig. 16 shows a case based on the axis L1 of the swash plate 31 not being offset relative to the axis L of the rotor 22, and in this case the shape of the contact point locus T on the expansion stroke side, which is the right-hand half of the figure, is symmetrical to that on the exhaust stroke side, which is the left-hand half thereof. In both a middle position of the expansion stroke and a middle position of the exhaust stroke, the contact point p between the spherical projection 61a of the piston 42 and the spherical depression 31a of the swash plate 31 is displaced to the inside from an axis L3 of the spherical depression 61a.

In particular, since in the expansion stroke the spherical projection 61a of the piston 42 driven by high-temperature, high-pressure steam is pressed strongly against the spherical depression 31a of the swash plate 31, if the contact point p between the spherical projection 61a and the spherical

depression 31a is offset from an axis L2 of the piston 42 or the axis L3 of the spherical depression 61a, a bending moment or a radially biased load acts on the piston 42, thus giving rise to the problems of an increase in the frictional resistance of sliding surfaces of the piston 42 and the cylinder sleeve 41 and the occurrence of abnormal wear.

However, in this embodiment, as shown in Fig. 15, since the axis L1 of the swash plate 31 is offset toward the exhaust stroke side (left-hand side in the figure) relative to the axis L of the rotor 22, the contact point locus T is offset toward the expansion stroke side (right-hand side in the figure), and it becomes possible to make the contact point p between the spherical projection 61a of the piston 42 and the spherical depression 31a of the swash plate 31 coincide with the axis L2 of the piston 42 and the axis L3 of the spherical depression 61a in the middle position of the expansion stroke. As a result, the piston 42 receives a load in a direction along the axis L2 of the piston 42 from the swash plate 31, thus reducing the bending moment and the radially biased load exerted on the piston 42 and thereby preventing any increase in the frictional resistance and any occurrence of abnormal wear. This reduction in the biased load exerted on the swash plate 31 by the piston 42 enables a contribution to be made to improvements in the durability of the swash plate 31 and the durability of the angular ball bearing 30 via which the swash plate 31 is supported by the swash plate holder 28.

When, as in this embodiment, the axis L1 of the swash plate 31 is offset toward the exhaust stroke side relative to the axis L of the rotor 22, it is possible to make the contact point p between the spherical projection 61a and the spherical depression 31a coincide with the axis L2 of the cylinder 42 and the axis L3 of the spherical depression 61a in the middle position of the expansion stroke, but in the middle position of the exhaust stroke the contact point p

between the spherical projection 61a and the spherical depression 31a is a great distance from the axis L2 of the cylinder 42 and the axis L3 of the spherical depression 61a (see Fig. 15). However, since in the exhaust stroke the load exerted on the piston 42 is small, the accompanying bending moment and radially biased load are very small and do not cause any particular problem.

In Fig. 17, the abscissa is a rotational angle of the rotor 22 measured from top dead center of the piston 42, and the ordinate is the bending stress acting on the piston 42 due to a reaction load from the swash plate 31. As is clear from this figure, in a region in which the speed of the piston 42 increases (the region where the angle from top dead center is 60° to 140°), that is, in a region where the lubrication conditions between the piston 42 and the cylinder sleeve 41 are most severe, the bending stress where there is offset, shown by the solid line, is smaller than the bending stress where there is no offset, shown by the broken line, and it can be seen that the effects of the present embodiment are usefully exhibited.

In contrast, in the region where the angle is 0° to 60°, which is immediately after top dead center, the bending stress where there is offset, shown by the solid line, is larger than the bending stress where there is no offset, shown by the broken line, but in this region the speed of the piston 42 is relatively small, the lubrication conditions are therefore mild, and there is no problem in practice.

The oil pump 49 provided on the output shaft 32 operates accompanying rotation of the rotor 22, and oil is taken in from the oil pan 21 via the oil pipe 52, the oil passage 15b of the front cover 15, and the intake port 53, discharged from the discharge port 54, and supplied to a space between the cylinder sleeve 41 and the small diameter part 62b formed in the middle part 62 of the

piston 42 via the oil passage 15c of the front cover 15, the oil passage 32a of the output shaft 32, the annular channel 32b of the output shaft 32, the oil holes 32c of the output shaft 32, the annular channel 41b of the cylinder sleeve 41, and the oil holes 41c of the cylinder sleeve 41. A portion of the oil retained by the small diameter part 62b flows into the spiral oil channels 62d formed in the middle part 62 of the piston 42 and lubricates the surface that slides against the cylinder sleeve 41, and another portion of the oil lubricates surfaces of the compression rings 66 and the oil ring 67 provided at the top part 63 of the piston 42 that slide against the cylinder sleeve 41.

Since water formed in the expansion chamber 43 by condensation of a portion of the supplied high-temperature, high-pressure steam inevitably enters between the sliding surfaces of the cylinder sleeve 41 and the piston 42 and contaminates the oil, the lubrication conditions of the sliding surfaces are severe, but by supplying a necessary amount of oil directly to the sliding surfaces of the cylinder sleeve 41 and the piston 42 from the oil pump 49 via the interior of the output shaft 32, it is possible to maintain a sufficient oil film, thereby ensuring the lubrication performance and enabling the dimensions of the oil pump 49 to be reduced.

Oil swept off the surface of the cylinder sleeve 41 that the piston 42 slides against by the oil ring 67 flows from the oil holes 63c formed in the base of the oil ring channel 63b into the hollow space 62a within the piston 42. The hollow space 62a communicates with the interior of the cylinder sleeve 41 via the plurality of oil holes 62c running through the middle part 62 of the piston 42, and the interior of the cylinder sleeve 41 communicates with the annular channel 41b on the outer periphery of the cylinder sleeve 41 via the plurality of oil holes 41c. Although the surroundings of the annular channel 41b are covered by the middle sleeve support flange 34 of the rotor 22, since the oil

hole 34b is formed in the sleeve support flange 34, the oil within the hollow space 62a of the piston 42 is biased radially outward due to centrifugal force, discharged to the space 68 within the heat-insulating cover 40 via the oil hole 34b of the sleeve support flange 34, and returned therefrom to the oil pan 21 via the oil holes 40a of the heat-insulating cover 40. During this process, since the oil hole 34b is positioned toward the axis L relative to the radially outer edge of the sleeve support flange 34, the oil that is present radially outside the oil hole 34b is retained in the hollow space 62a of the piston 42 by centrifugal force.

In this way, the oil retained in the hollow space 62a within the piston 42 and the oil retained in the small diameter part 62b on the outer periphery of the piston 42 are supplied from the small diameter part 62b to the top part 63 side during the expansion stroke in which the volume of the expansion chamber 43 increases, and are supplied from the small diameter part 62b to the end part 61 side during the exhaust stroke in which the volume of the expansion chamber 43 decreases, and it is therefore possible to ensure reliable lubrication over the entire axial region of the piston 42. Furthermore, as a result of the oil flowing within the hollow space 62a of the piston 42, the heat of the top part 63, which is exposed to high-temperature, high-pressure steam, is transmitted to the end part 61, which has a low temperature, and it is thus possible to avoid the temperature of the piston 42 increasing locally.

When high-temperature, high-pressure steam is supplied from the fourth steam passage P4 to the expansion chamber 43, since the heat-insulating space 65 is formed between the middle part 62 and the top part 63 of the piston 42, which faces the expansion chamber 43, and the heat-insulating space 70 is formed in the rotor head 38, which faces the expansion chamber 43, it is possible to minimize the escape of heat from the expansion chamber 43 to the

piston 42 and the rotor head 38, thereby contributing to an improvement in the performance of the expander M. Furthermore, since the large volume hollow space 62a is formed within the piston 42, not only is it possible to reduce the weight of the piston 42, but it is also possible to reduce the heat capacity of the piston 42, thereby enabling the escape of heat from the expansion chamber 43 to be suppressed yet more effectively.

Since the expansion chamber 43 is sealed by interposing the metal gasket 36 between the rear sleeve support flange 35 and the rotor head 38, in comparison with a case in which the expansion chamber 43 is sealed via a thick annular seal, unnecessary volume around the seal can be reduced, thus ensuring that the expander M has a large volume ratio (expansion ratio) and thereby improving the thermal efficiency, which enables the output to be increased. Moreover, since the cylinder sleeve 41 is formed separately from the rotor 22, the material of the cylinder sleeve 41 can be selected without being restricted by the material of the rotor 22, while taking into consideration the thermal conductivity, heat resistance, strength, abrasion resistance, etc., and, moreover, it is possible to replace only a worn or damaged cylinder sleeve 41, which is economical.

Furthermore, since the outer peripheral face of the cylinder sleeve 41 is exposed through the two cutouts 57 and 58 formed circumferentially in the outer peripheral face of the rotor 22, not only is it possible to reduce the weight of the rotor 22, but it is also possible to reduce the heat capacity of the rotor 22, thereby improving the thermal efficiency and, moreover, the cutouts 57 and 58 function as a heat-insulating space, thus suppressing the escape of heat from the cylinder sleeve 41. Furthermore, since the outer peripheral part of the rotor 22 is covered by the heat-insulating cover 40, it is possible to suppress the escape of heat from the cylinder sleeve 41 yet more effectively.

Since the rotary valve 71 supplies and discharges steam to and from the axial piston cylinder group 56 via the flat sliding surfaces 77 between the stationary valve plate 73 and the movable valve plate 74, it is possible to prevent the leakage of steam effectively. This is because the flat sliding surfaces 77 can easily be machined with high precision, and control of the clearance is easy compared with cylindrical sliding surfaces. Moreover, since a surface pressure is generated on the sliding surfaces 77 of the stationary valve plate 73 and the movable valve plate 74 by applying a preset load to the valve main body 72 by means of the plurality of preload springs 85, it is possible to suppress the leakage of steam past the sliding surfaces 77 yet more effectively.

Furthermore, since the valve main body 72 of the rotary valve 71 is made of stainless steel, which has a large coefficient of thermal expansion, and the stationary valve plate 73 fixed to the valve main body 72 is made of carbon or a ceramic, which has a small coefficient of thermal expansion, there is the possibility that the centering between the two might be displaced due to a difference in the coefficients of thermal expansion, but since the fixing ring 79 is fixed to the valve main body 72 by means of the plurality of bolts 80 in a state in which the step 79a on the inner periphery of the fixing ring 79 is press-fitted over the outer periphery of the stationary valve plate 73 in a spigot joint manner and the step 79b on the outer periphery of the fixing ring 79 is press-fitted over the outer periphery of the valve main body 72 in a spigot joint manner, it is possible to carry out precise centering of the stationary valve plate 73 relative to the valve main body 72 by the aligning action of the press-fitting of the fixing ring 79 and prevent the timing of supply and discharge of steam from deviating, thereby preventing deterioration in the performance of the expander M. Moreover, it is possible to make the abutting surfaces of the stationary valve plate 73 and the valve main body 72 come into intimate and uniform contact by

virtue of the securing force of the bolts 80, thereby suppressing the leakage of steam past the abutting surfaces.

Moreover, since the rotary valve 71 can be attached to and removed from the casing main body 12 merely by removing the rear cover 18 from the casing main body 12, the ease of maintenance operations such as repair, cleaning, and replacement can be greatly improved. Furthermore, although the rotary valve 71 through which the high-temperature, high-pressure steam passes reaches a high temperature, since the swash plate 31 and the output shaft 32, where lubrication by oil is required, are disposed on the opposite side of the rotor 22 to the rotary valve 71, degradation of the lubrication performance of the swash plate 31 and the output shaft 32 due to heating of the oil by the heat of the rotating valve 71, which reaches a high temperature, can be prevented. Moreover, the oil also exhibits the function of cooling the rotary valve 71, thus preventing overheating.

Although an embodiment of the present invention is explained above, the present invention can be modified in a variety of ways without departing from the spirit and scope thereof.

INDUSTRIAL APPLICABILITY

As hereinbefore described, the present invention can be applied suitably to the expander M of the Rankine cycle system, but it can be applied to an expander for any purpose as long as it is an expander equipped with an axial piston cylinder group.